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CERTIFICATE

This certificate is issued in support of an application for Patent registration in a country outside New Zealand pursuant to the Patents Act 1953 and the Regulations thereunder.

I hereby certify that annexed is a true copy of the Provisional Specification as filed on 6 September 2002 with an application for Letters Patent number 521263 made by KENNETH WILLIAM PATTERSON DRYSDALE.

Dated 16 September 2003.

Neville Harris

Neville Harris
Commissioner of Patents, Trade Marks and Designs

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Patents Form No. 4 Our Ref: CC504085

Patents Act 1953

PROVISIONAL SPECIFICATION

APPARATUS, METHOD AND SOFTWARE FOR USE WITH AN AIR CONDITIONING CYCLE

We, **KENNETH WILLIAM PATTERSON DRYSDALE,** an Australian citizen of 8A Elm Avenue, Belrose, New South Wales 1640, Australia, do hereby declare this invention to be described in the following statement:

PT0406731

Apparatus, Method and Software For Use With An Air Conditioning Cycle

Technical Field

The present invention relates to thermodynamic cycles and the control of thermodynamic cycles and more particularly, but not exclusively to the control of an air conditioning system.

Background

Air conditioning systems have become a huge draw on electricity power in many of the major cities of the world and are viewed as an essential component of many large buildings in order to maintain a level of environmental control within the building. At the same time as air conditioning systems continue to increase in number, it is becoming increasingly recognised that electricity is a limited resource and in some places demand is exceeding supply or is forecast to in the near future.

It has become important to identify potential areas for saving in electricity consumption. If any savings can be made in air conditioning systems, then there is potential to make an overall huge saving in the consumption of electricity.

It is therefore an object of the present invention to provide a control system that may be used to provide at least some savings in electricity consumption for an air conditioning system or at least to provide a control system that will provide the public with a useful alternative.

Further objects of the present invention may become apparent from the following description.

Summary of the Invention

According to one aspect of the present invention, there is provided a control system for a thermodynamic cycle including a compressor, the control system including:

- sensing means for providing a measure of an output of the thermodynamic cycle;
- control means for the compressor, wherein the control means is in communication with said sensing means to receive as inputs said measure of an output of the thermodynamic cycle and a measure of the work input of the compressor;

wherein the control means is operable to compute a measure of efficiency from said inputs and vary the speed of the compressor to maximise said measure of efficiency or to maintain said measure of efficiency at a predetermined level.

Preferably, the control system further includes second control means for a TX valve or equivalent and sensing means for providing a measure of the temperature of a controlled area, wherein the second control means receives as a further input said measure of the temperature of a controlled area, and is operable to open or close the TX valve or equivalent in response to sensed variations in temperature in the controlled area in relation to a target measure.

Preferably, the second control means may further receive as an input a measure indicative of the amount of refrigerant in the cycle which is vaporised after an evaporation phase in the cycle and to open or close the TX valve or equivalent to maintain vaporised refrigerant after the evaporation phase.

Preferably, the operation of the second control means to maintain vaporised refrigerant after the evaporation phase is performed after a predetermined delay from the control means opening or closing the TX valve in response to said sensed variations of temperature.

Preferably, the control system may include third control means for a condenser in the thermodynamic cycle, the control system varying the operation of the condenser to maintain a required level of cooling of refrigerant by the condenser.

Preferably, the control means, second control means and third control means may be a single microcontroller or microprocessor or a plurality of microcontrollers or microprocessors with at least selected microcontrollers or microprocessors in communication with each other to allow management of the timing of the functions of the control system.

According to another aspect of the present invention, there is provided a computer program or a computer program product storing a computer program having instructions to cause one or more microcontrollers or microprocessors to operate as the control means, second control means and third control means described in the immediately preceding paragraphs.

According to a further aspect of the invention, there is provided a thermodynamic cycle for a refrigerant, the cycle including a compressor, TX valve or an equivalent and a condenser, wherein the compressor is controlled by the control means described in the preceding paragraphs.

Preferably, the TX valve or equivalent may controlled by the second control means described in the preceding paragraphs.

Preferably, the condenser may be controlled by the third control means described in the preceding paragraphs.

Preferably, the thermodynamic cycle may be arranged with the compressor upstream of the TX valve or equivalent, the TX valve or equivalent located upstream of an evaporator and a condenser located between said evaporator and said compressor.

Preferably, the thermodynamic cycle may not include a receiver following the condenser.

Preferably, the thermodynamic cycle may include one or more turbines in the cycle, wherein energy created by rotation of the turbines is fed into a power supply to the compressor.

According to another aspect of the present invention, there is provided a control system for a thermodynamic cycle substantially as herein described with reference to the accompanying drawings.

According to a further aspect of the present invention, there is provided a thermodynamic cycle incorporating a control system substantially as herein described with reference to the accompanying drawings.

Further aspects of the present invention may become apparent from the following description, given by way of example only and with reference to the accompanying drawings.

Brief Description of the Drawings

Figures 1 and 2: show a flow chart of the main control process for a thermodynamic cycle according to an aspect of the present invention.

Figure 3: shows a flow chart of a control process for adjusting the temperature output of a thermodynamic cycle through variation of a TX valve or equivalent according to another aspect of the present invention.

Figure 4: shows a flow chart of a possible start subroutine for the control process shown in Figures 1 to 3.

Figure 5: shows a flow chart of a possible time delay subroutine for the control process shown in Figures 1 to 3.

Figure 6: shows a flow chart of a control process for a condenser according to another aspect of the present invention.

Figure 7: shows a flow chart of a subroutine to check the status of a master switch for the control process of the present invention.

Figure 8: shows a representation of a first thermodynamic cycle according to an embodiment of the present invention.

Figure 9: shows a representation of a second thermodynamic cycle according to another embodiment of the present invention.

Detailed Description of the Drawings

Referring to the accompanying Figures 1 to 9, a series of flow diagrams illustrating an example of the computational process of the present invention that may be performed to control an air conditioning cycle is shown. The process may be controlled by any suitable

microcontroller, microprocessor or similar having a control output to control the drive signal of a motor controller for a compressor.

Referring first to Figures 1 and 4, the microcontroller after power up performs an initialisation subroutine. It is anticipated that whether or not the below described optimisation process is performed will be controlled by a single master switch SW1, which may be a manual input switch. If the switch SW1 is in the off position, the algorithm continues to check the switch until it is detected as being in the on position. Referring particularly to Figure 4, if the optimisation algorithm is to be invoked, selected flags, registers and counters may be initialised, typically by setting to zero if this is required for the particular implementation of the algorithm.

The micro controller receives as inputs the target co-efficient of performance COP_1, the required motor speed increment K_2 for the compressor and an air conditioning refrigerant constant K_1. K_1 may be determined experimentally for the particular air conditioning cycle. These inputs may be entered manually through a user interface, such as a keypad. Alternatively, these inputs may be set at the time of programming of the microcontroller.

In addition, the microcontroller receives as inputs the temperature of the air flowing into the evaporator T_1, the temperature of the air leaving the evaporator T_2 and the compressor motor power KW_1. Having received these inputs, the microcontroller then computes the co-efficient of performance COP_2 according to the equation:

$$COP_2 = K_1|T_1-T_2| / W$$

Other measures relating the output of the cycle to the compressor work input may be used if required. As herein described, the presently contemplated preferred embodiment uses measures of temperature difference to provide a measure of the useful heat transferred by the system, as temperature measurements may be relatively easily obtained. However, alternative measures of system performance may be used that relate the system output to the compressor input.

The computed co-efficient of performance COP_2 is then compared to the target, COP_1. If the value of the target COP_1 is less than COP_2, the compressor speed is increased by K_2. Conversely, if the target COP_1 is greater than the computed COP_2, the motor speed is decreased by K_2. A delay subroutine is then executed to allow for any lag

in the response of the cycle to the change in compressor speed. The required time delay can be determined experimentally by forcing adjustments of the compressor speed by increments of K_2 and measuring the maximum time for the air conditioning cycle to return to steady state conditions. Any suitable delay subroutine may be used to effect this delay. This brings the algorithm to point 1 as indicated in Figure 1. After the delay subroutine has been executed, the next step in the control part of the algorithm is indicated by 2 in Figure 2. Throughout the description of the algorithm, the flags such as 1 and 2 are meant to be followed consecutively.

Figure 5 shows a possible delay subroutine in more detail. The delay may receive a clock signal RTC_1, and count a predetermined number of cycles before exiting the routine. The counter may increment by a manually entered value DEL_1 until a predetermined value RTC is reached. Upon exit, the microcontroller may perform each of the processes commencing with flags 2, 4, 6 and 8, see Figures 2, 3, 6 and 7 respectively.

Referring to Figure 2, after the compressor speed has been incremented, decremented or maintained at the same level, the microcontroller receives as inputs the temperature of the air exiting the evaporator T_2, a measurement of the status of a TX valve or equivalent TX_1, a set increment for varying the operation of the TX valve and a set point SP_1 indicating the required temperature in the control area of the air conditioning system. If the temperature T_2 is less than the set point, the TX valve is closed and if the temperature T_2 is greater than the set point, the TX valve is opened by increment K_3. Otherwise, the TX valve is maintained in its current position. The microcontroller invokes a second delay subroutine in order to allow the cycle to reach a steady state or near steady state. The cycle then repeats, constantly reading and computing the actual COP with the target COP and adjusting the compressor speed and constantly comparing the output temperature with the set point temperature and adjusting the TX valve to compensate for variations.

With variation of the TX valve setting, it becomes desirable to check that the TX valve is still operating so that the refrigerant in the suction line of the compressor after the evaporator is sufficiently super heated to be at the vapour state. Therefore, each time when the delay subroutine following variation of the TX valve is invoked, the microcontroller performs an additional check on the operation of the TX valve. This is shown in Figure 3.

Based on prior knowledge of the system, the temperature differential target $TDIFF_1$ is set to ensure that the refrigerant inputted to the compressor is in the vapour state. The micro controller also receives as inputs the reading TX_1, the evaporator output temperature T_2, the refrigerant temperature immediately after the evaporator T_3 and a TX valve increment K_4. K_4 may be dependent on the compressor speed in order to maintain temperature changes despite the change in flow rate of the refrigerant around the air conditioning cycle dependent on the compressor speed. The actual temperature difference between the evaporator environmental output and the evaporator refrigerant output is compared to the set point and if the actual temperature is less than the set point temperature, the TX valve is opened by increment K_4 and if the calculated temperature difference is greater than the set point, the TX valve is closed. A further delay subroutine is invoked to accommodate the time lag in the cycle.

With variations in the compressor speed and TX valve opening, the operation of the condenser will also vary. Therefore, the controller also controls the drive fan to the condenser. This process is shown in Figure 6. The target temperature TT_1 and incremental speed variation for the condenser K5 are manually entered and the system automatically measures the temperature of the refrigerant prior to entering the receiver T_4 and the actual fan speed CFS1. If T_4 is greater than TT_1, the fan speed is increased, thereby further cooling the working fluid in the condenser. If T_4 is less than TT_1, the fan speed is decreased, removing less heat from the working fluid as it passes through the condensor. A further time delay is invoked after variation of the condenser fan operation.

The controller may also periodically sense whether the switch SW_1 has changed state. This may be performed during each time delay sub routine, as indicated in Figure 7.

Figure 8 shows an air conditioning/refrigeration cycle, generally referenced by arrow 100, according to another aspect of the present invention. The above described control methodology may be applied to the cycle 100, with suitable temperature sensors provided in the cycle.

The cycle 100 may differ from air conditioning or refrigeration cycles of the prior art in that the TX valve and receiver common to the cycles of the prior art, have been replaced by a turbine 114 located between the condenser 105 and evaporator 122. A suitable turbine is described in the applicant's copending New Zealand patent application

number 517088, which is hereby incorporated herein in its entirety where appropriate. An optional thermoelectric generator 103 may precede the condenser 105.

The cycle 100 is described below in terms of its preferred form as presently contemplated, wherein the working fluid may be accelerated to sonic and/or supersonic velocities in various parts of the cycle. Those skilled in the art will recognise that the cycle may be adapted to for use with subsonic velocities throughout by use of appropriately shaped nozzles and diffusers as are well known to the art.

High pressure working fluid may exit a compressor discharge line 102 in a substantially vapour phase and may enter either an optional thermoelectric generator 103 or may pass straight to a condenser 105. The thermoelectric generator 103, if present, may produce a low voltage DC output 103a which may be converted to a high voltage output 104a through a DC to DC converter 104.

The condenser 105 may remove heat from the working fluid. The amount of heat rejected may be controlled by the speed of a condenser fan 106 which blows air over the condenser 105. The speed of the condenser fan 106 may be determined by a variable speed drive 107, controlled by a master variable speed drive 109 through a communications link 108. The variable speed drive 107 includes suitable software to control the speed of the condenser fan 106.

References to T_5, T_6, T_7 and T_8 refer to the temperature of the environment outside the air conditioned area, the temperature of the working fluid at the compressor discharge, the air temperature of the heated air exiting the condenser coil and the temperature of the working fluid exiting the condenser respectively.

The master variable speed drive 109 may include thermocouple inputs 110, 111, 112 and 113 to provide information on temperatures T_5, T_6, T_7 and T_8 respectively, which may enable the master variable speed drive 109 to determine the heat rejected from the condenser 105 and the density of the working fluid entering the turbine 114.

A mass flow sensor 115 may measure the mass flow rate of the working fluid entering the turbine 114.

By measuring the mass flow rate of the working fluid and temperatures T_5 to T_8, the software in the master variable speed drive 109 may estimate the density of the working fluid entering the turbine 114 and may adjust the speed of the compressor 101 and/or condenser fan 106 and/or evaporator fan 126 to ensure that it is sufficiently low that the vapour passing through the throat of a converging/diverging nozzle 117, which feeds the turbine 114, is at a substantially sonic velocity.

The sonic working fluid exiting the turbine nozzle throat may continue to accelerate in a diverging section of the nozzle until it reaches a supersonic velocity.

The high velocity working fluid drives the turbine rotor 119 including turbine blades 120 or other blade structure such as a channel blade structure 120a. The turbine may drive a load 121, for example an electric generator, via a suitable coupling 120.

Acceleration of the working fluid within the nozzle, preferably to sonic or supersonic velocities, may cause a fall in its temperature and pressure. Energy may then be removed from the working fluid as a result of the flowing through the turbine 114.

A mixture of high velocity low pressure working fluid in both vapour and liquid phases is passed into an evaporator 122 which may preferably be designed to sustain either sonic or supersonic flow within, and thereby keep the working fluid within the evaporator 122 at a sufficiently low temperature and pressure to sustain a heat exchange from the environment to the evaporator coil 123.

The evaporator coil 123 may absorb heat from the warmer air 124 outside the evaporator 122. The cooled air 125 may be removed from the evaporator 122 by an evaporator fan 126.

As the working fluid is heated within the evaporator 122 it's velocity within the evaporator coil 123 may be reduced to close to a sonic velocity.

The speed of the evaporator fan 126 may be varied by a further variable speed drive 130 connected to the power input of the evaporator fan 126 and controlled by the master variable speed drive 109 through a communications link 108a.
The speed of the evaporator fan 126 may be varied in response to the drop in temperature of the air 124 flowing over the evaporator 122.

After passing through the evaporator coil 123, the high velocity working fluid may pass through a diffuser 127 at the entrance to an accumulator 128, which, in the preferred embodiment, may reduce the velocity of the fluid to below sonic.

The accumulator 128 may ensure that any remaining liquid phase fluid is evaporated prior to entering the compressor input 129. The accumulator 128 may also act as a working fluid reservoir to replace the receiver used by some air conditioning/ refrigeration cycles of the prior art.

The master variable speed drive 109 may control the speed of the compressor 101 to optimise it's COP, substantially as described above, although the TX valve control will be omitted due to the elimination of the TX valve from the cycle 100.

If the turbine 114 is driving an electrical generator 121 then the electrical generator 121 may be either of the DC or AC type. Preferably the generator 121 may be a high voltage DC generator of the order of 670 volts output. In the preferred case the DC power output 114b may be coupled into the DC bus bar 109b of the master variable speed drive 109 through a diode and capacitor isolation circuit which may only allow power to flow in one direction, thus avoiding any feedback of mains power 150 to the generator 121.

Referring next to Figure 9 an air conditioning/refrigeration cycle according to a still further aspect of the present invention is generally referenced 200.

The air conditioning cycle 200 may vary from air conditioning cycles of the prior art in that the TX valve and condenser have been replaced by a turbine 204 and, optionally, a thermoelectric generator 203.

The air conditioning cycle 200 operates as follows:

High pressure working fluid vapour exits the discharge line 202 of a compressor 201. The working fluid from the discharge line 202 may pass through an optional thermoelectric generator 203, in which an electric current may be produced, and may then flow to a suitable turbine 204. Alternatively the vapour may flow directly from the compressor to the turbine 204.

If present, the current from the thermoelectric generator 203 may be passed through a DC to DC converter 203a to raise the voltage to a suitable level for combining the power output 203b with power produced by one or more of the other power generation components as described below.

The high pressure vapour may be accelerated, preferably by a converging/diverging nozzle 205 and may acquire an elevated, preferably sonic, velocity at the nozzle throat.

Preferably, the expanding vapour in the diverging section of the nozzle 205 may acquire a supersonic velocity prior to entering the turbine 204.

The working fluid exiting the nozzle throat may continue to accelerate in the diverging section until it reaches supersonic velocities, and may then drive a turbine rotor coupled to a load 209, for example an electrical generator.

In accelerating, the pressure and temperature of the working fluid may fall. Energy is removed from the working fluid through the energy exchange process as a result of passing through the turbine 204.

The working fluid may now be a mix of relatively high velocity, low pressure vapour and liquid and is passed into an evaporator 215, which may be adapted to sustain the velocity of the fluid and thereby keep the temperature and pressure of the working fluid within the evaporator 215 low enough to sustain a heat exchange from the ambient air 214.

Heat may be exchanged from the relatively warm ambient air 214 to the high velocity working fluid within the evaporator 215. The cooled ambient air 217 may be removed from the evaporator 215 by a fan 216.

As the working fluid is heated by the warm ambient air 214 it's velocity within the evaporator 215 is reduced, possibly to close to a sonic velocity.

After passing through the evaporator 215 the working fluid is passed through a diffuser 221 which reduces the speed of the working fluid, preferably to subsonic, prior to the working fluid entering an accumulator 222.

The accumulator 222 ensures that any liquid phase working fluid is evaporated prior to entering the compressor input 223, thereby completing the cycle.

Those skilled in the art will recognise that the accumulator 222 may act as a working fluid reservoir to replace the receiver in a conventional air conditioning/ refrigeration cycle.

The thermoelectric generator output 203b may be connected to the master variable speed drive 210.

The master variable speed drive 210 may ensure that the input power 211 drawn is reduced by the sum of power provided by the thermoelectric power output 203b and the turbine power output 204b, thereby realizing an energy saving and associated cost saving.

As described above, temperature measurements T_1 and T_2 refer to the ambient temperature of the warm ambient temperature air 214, blown onto evaporator coil 215 by an evaporator fan 216, and the cold air 217 exiting from the evaporator coil 215 respectively.

The master variable speed drive 210 may be connected to the compressor power input 211 and may include thermocouple inputs 212 and 213 to provide inputs relative to temperature measurements T_1 and T_2 respectively, which enable the master variable speed drive 210 to determine the desired speed of the compressor 201 and thus produce the optimum COP for the system, as previously described.

A variable speed drive 218 is controlled by the master variable speed drive 210 through a communications link 219, and is connected to a power input 220 of the evaporator fan 216 to modulate the evaporator fan speed in response to thermocouple inputs 212 and 213.

The master variable speed drive 210 may have an additional thermocouple input 202A proportional to the temperature T_{11} of the fluid in the discharge line 202, which is in turn dependant on the pressure of the fluid in the compressor discharge line 202. The thermocouple input 202A may therefore be used by the software in the master variable

speed drive 210 to indirectly determine the pressure in the discharge line 202, so that if there is heat imbalance in the system which causes the discharge temperature T_{11} to rise, indicating an increase in the pressure in the discharge line 202, then the software is alerted and can modulate the speed of the evaporator fan 216 via the variable speed drive 218, to cause less heat to be acquired from evaporator 215 until the heat imbalance is corrected.

Those skilled in the art will recognise that the air conditioning cycles described above may be more energy efficient than those of the prior art, due to energy recovered by the turbine and, where used, the thermoelectric generator, as well as the control of the compressor speed to optimize the overall Coefficient of Performance.

Where in the foregoing description reference has been made to specific components or integers having known equivalents, then such equivalents are hereby incorporated herein as if individually set forth.

The foregoing description of the invention has been given by way of example only and with reference to preferred embodiments as presently contemplated. Those skilled in the relevant arts will appreciate that modifications or improvements may be made to the invention without departing from its scope.

COEFFICIENT OF PERFORMANCE (COP) OPTIMISATION ALGORITHM
(SPEED CONTROL OF COMPRESSOR)

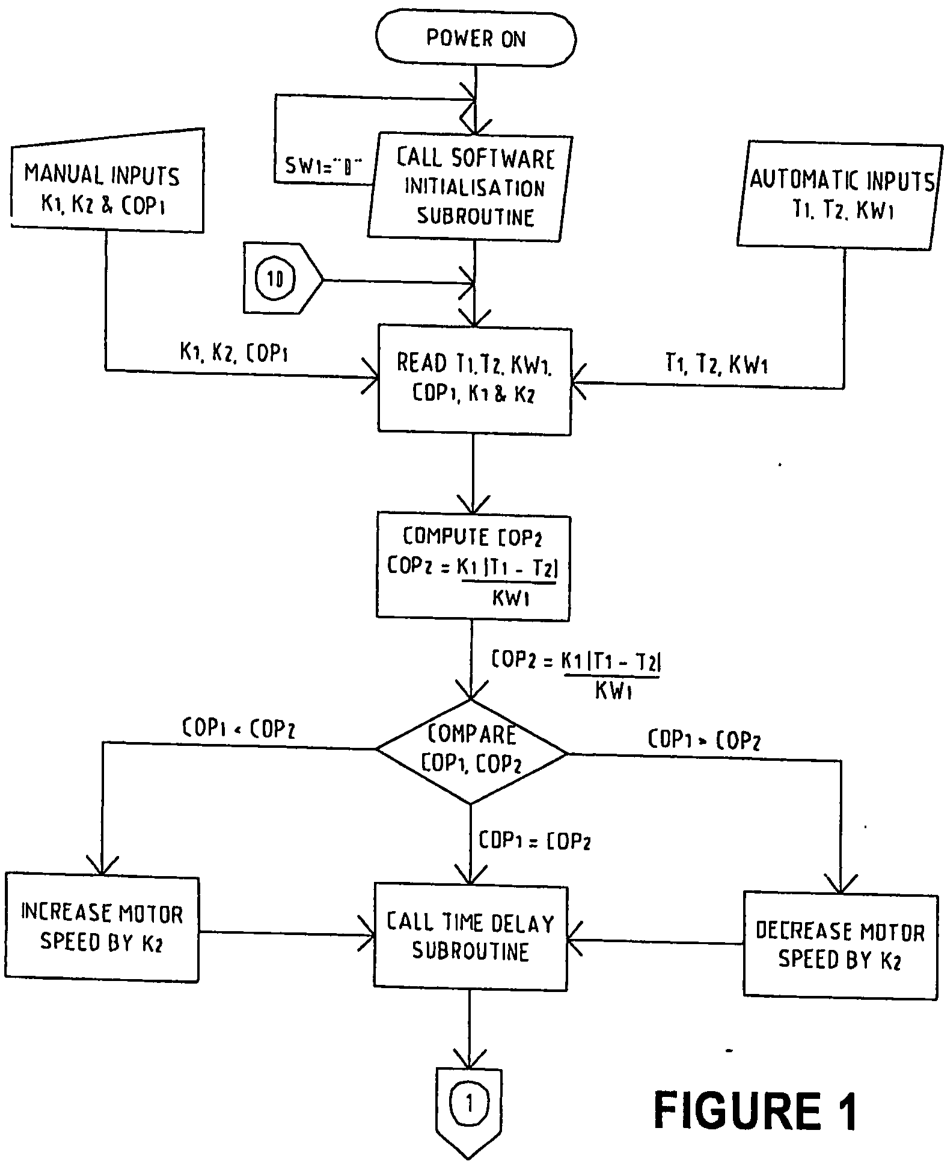


FIGURE 1

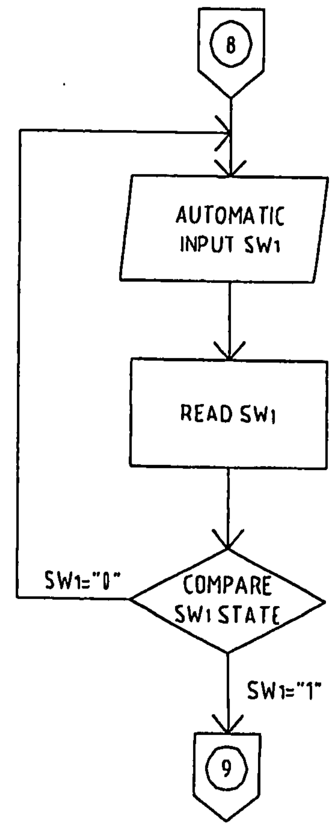


FIGURE 7

COP OPTIMISATION ALGORITHM CONTINUED
(SET POINT CONTROL OF AIR OFF THERMAL LOAD)

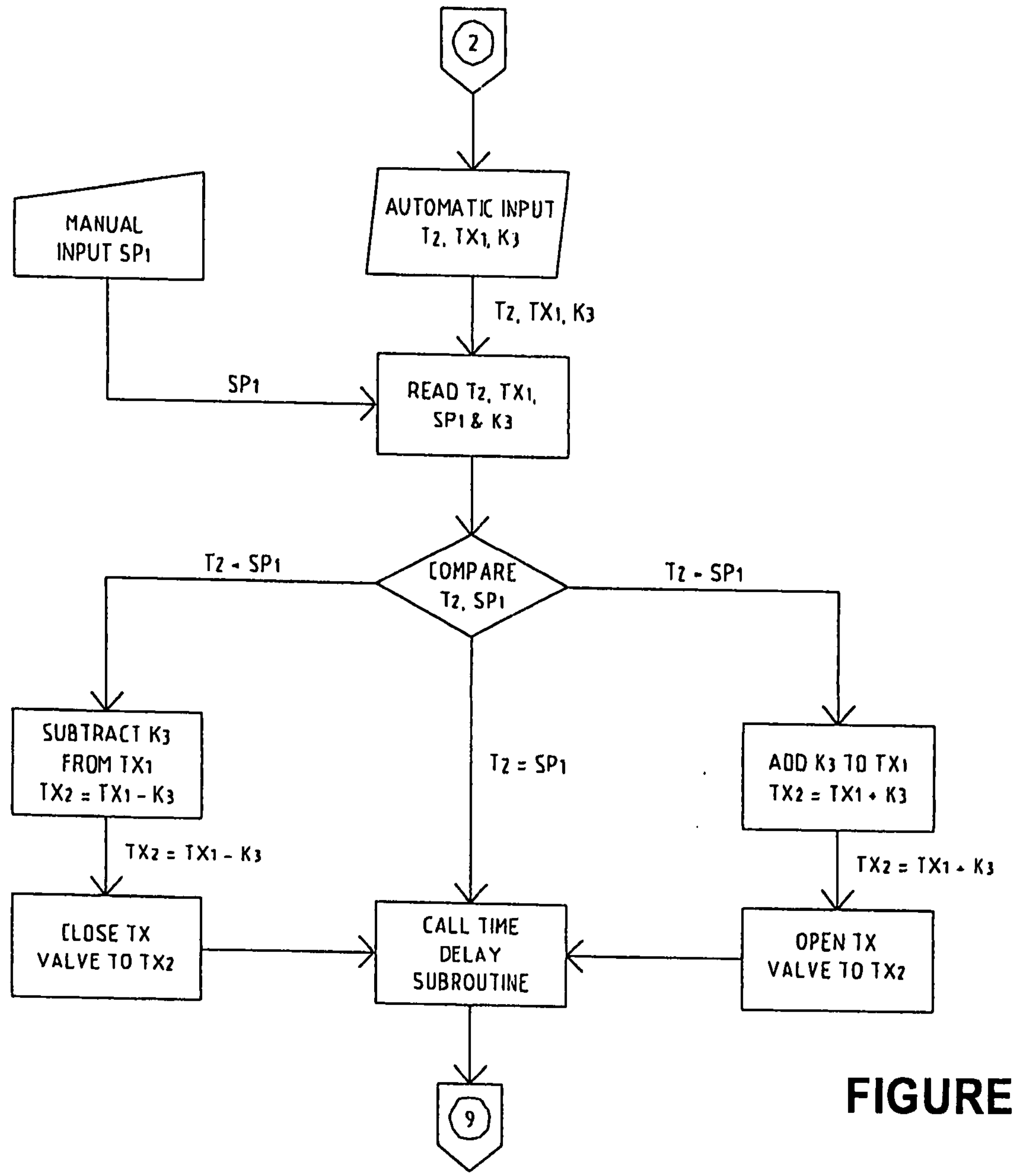


FIGURE 2

COP OPTIMISATION ALGORITHM CONTINUED
(SUPERHEAT CHECK ON TX VALVE SETTING)

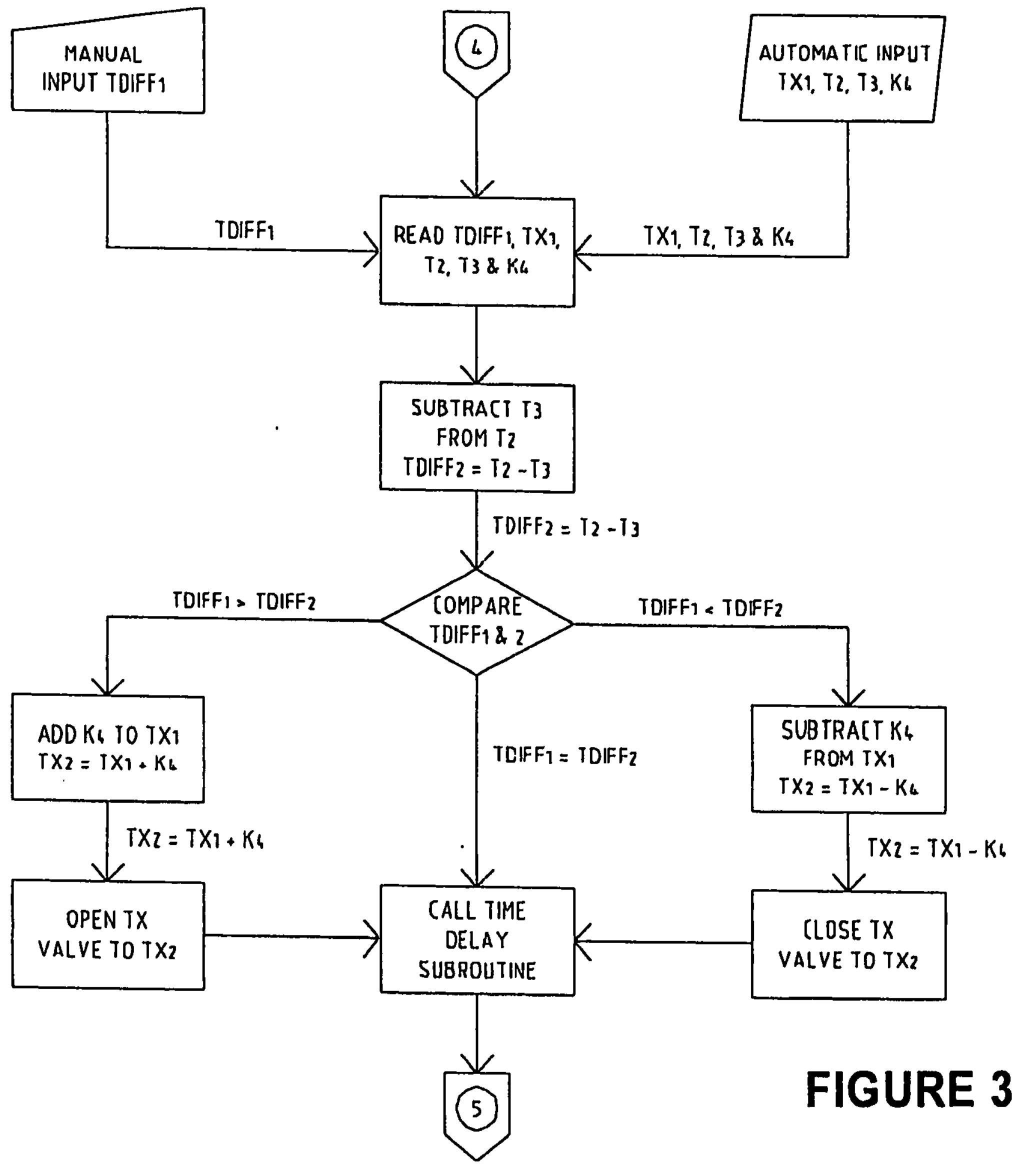


FIGURE 3

COP OPTIMISATION ALGORITHM CONTINUED
(HARDWARE/SOFTWARE INITIALISATION SUBROUTINE)

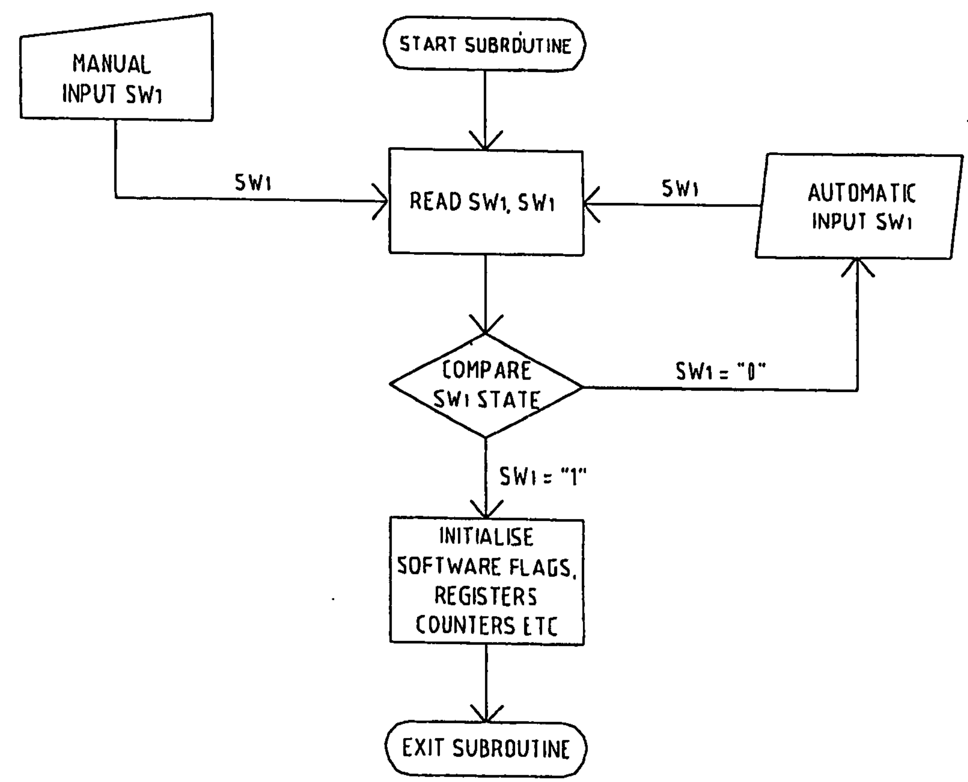


FIGURE 4

COP OPTIMISATION ALGORITHM CONTINUED
(TIME DELAY SUBROUTINE)

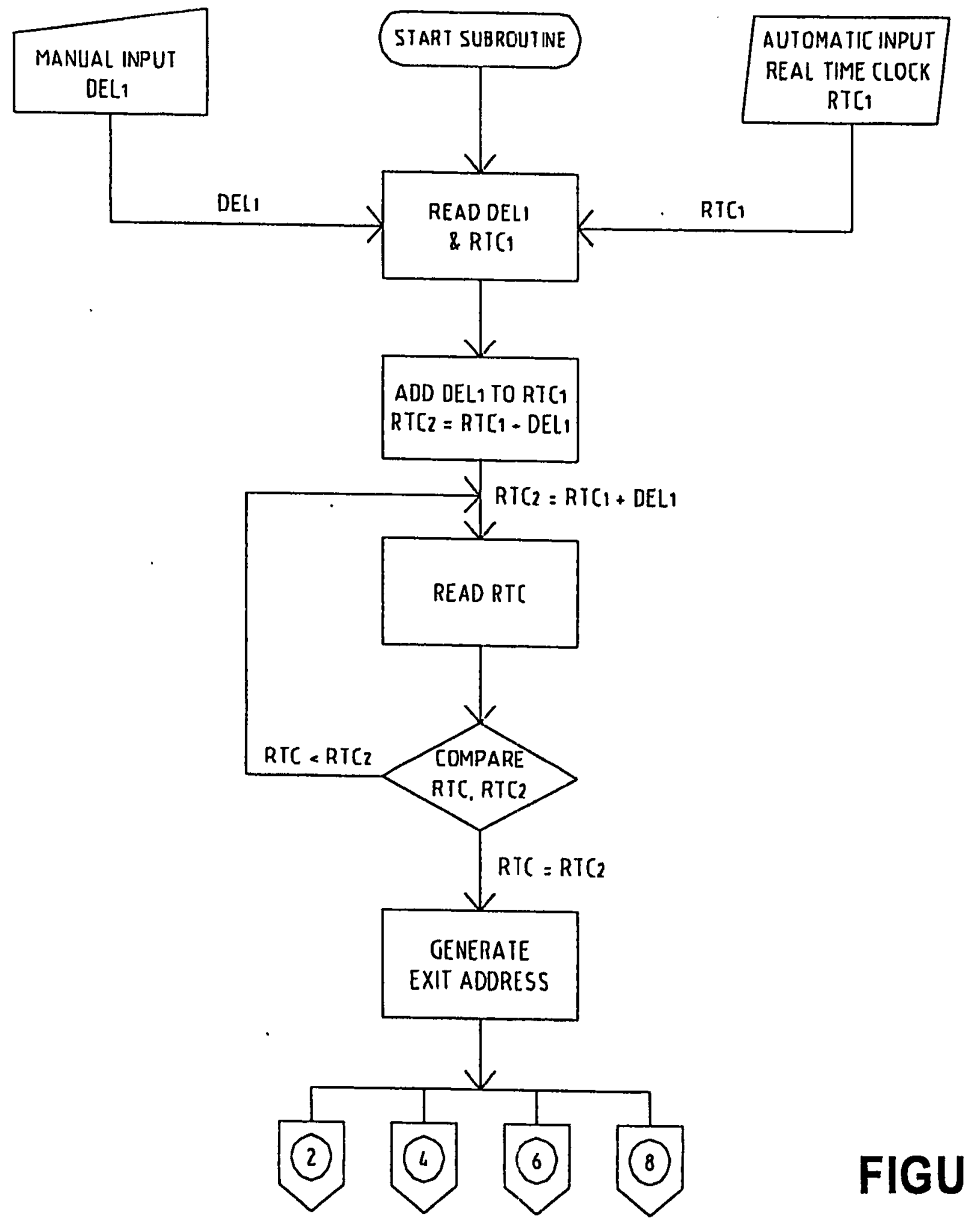


FIGURE 5

COP OPTIMISATION ALGORITHM CONTINUED
(CONDENSER FAN CONTROL)

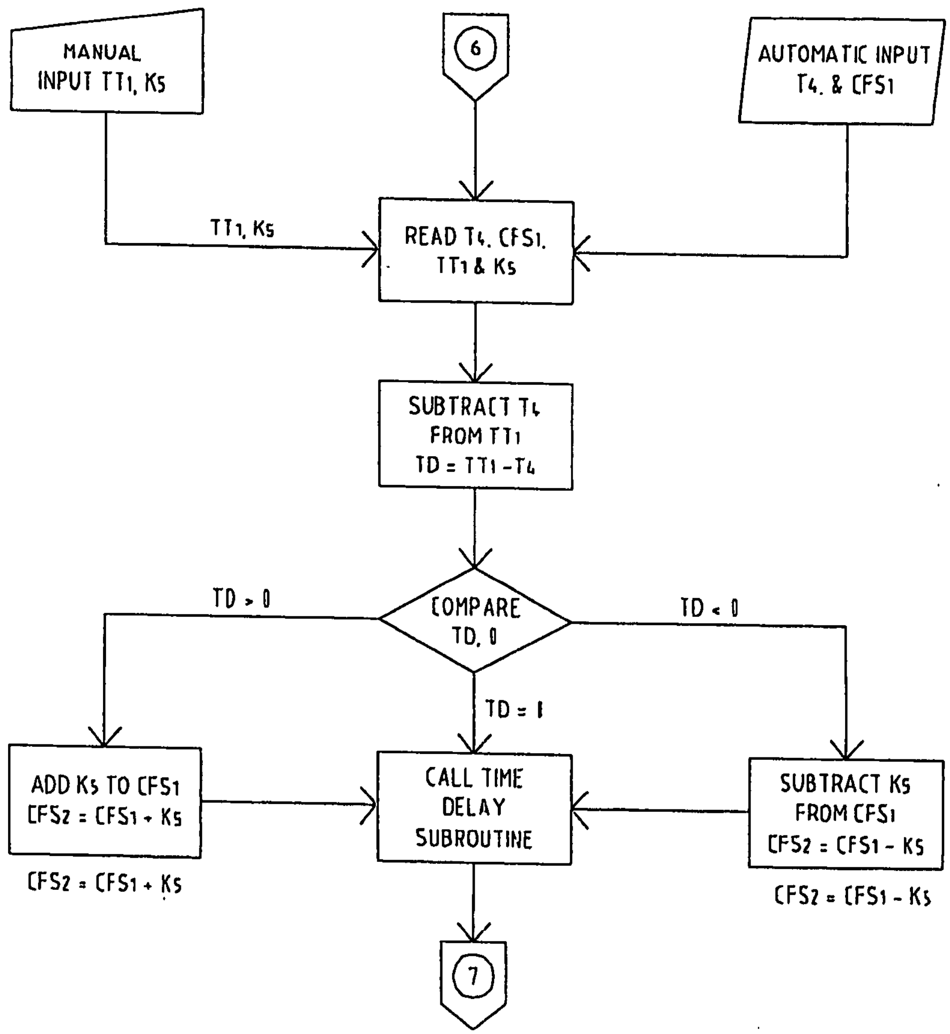
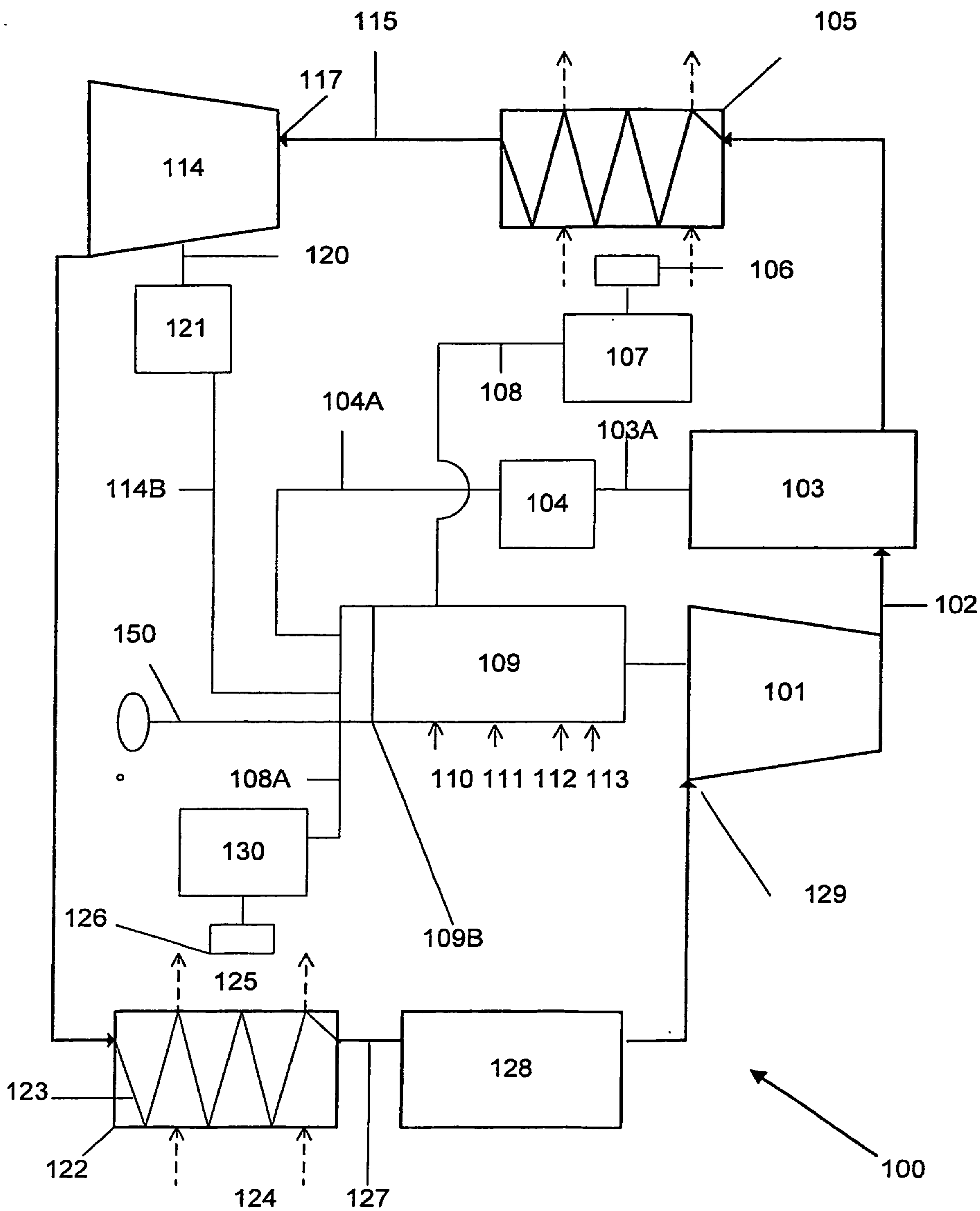


FIGURE 6



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108A
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129
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109B
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FIGURE 8

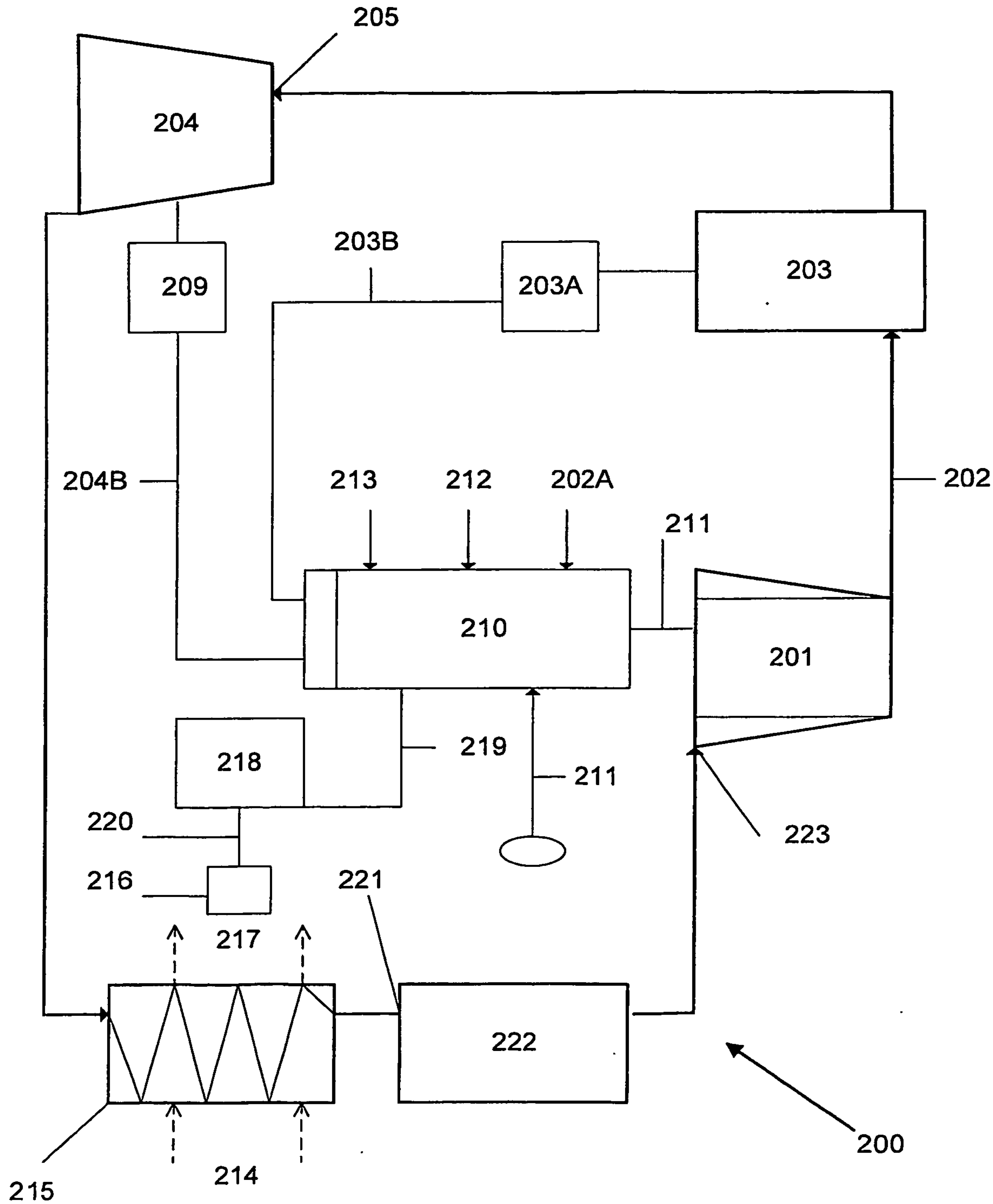
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FIGURE 9